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Pat/GB2005/002999.



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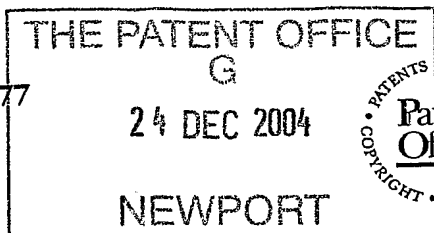


*L. Mahoney*

Signed

Dated 31 August 2005





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24 DEC 2004

1. Your reference

P38572-INGR/GMU

2. Patent application number

(The Patent Office will fill this part in)

0428343.8

3. Full name, address and postcode of the or of each applicant (underline all surnames)

Pursuit Dynamics plc  
Unit 1, Anglian Business Park  
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Patents ADP number (if you know it)

If the applicant is a corporate body, give the country/state of its incorporation

United Kingdom

08333072002

4. Title of the invention

"Method and Apparatus for Moving a Fluid"

5. Name of your agent (if you have one)

Murgitroyd & Company

"Address for service" in the United Kingdom to which all correspondence should be sent (including the postcode)

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Glasgow  
G5 8PL

Patents ADP number (if you know it)

1198013

1198015

6. Priority: Complete this section if you are declaring priority from one or more earlier patent applications, filed in the last 12 months.

Country

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Date of filing  
(day / month / year)

7. Divisionals, etc: Complete this section only if this application is a divisional application or resulted from an entitlement dispute (see note f)

Number of earlier UK application

Date of filing  
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Yes

Answer YES if:

- a) any applicant named in part 3 is not an inventor, or
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Continuation sheets of this form	-
Description	46
Claim(s)	-
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Priority documents	-
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11. I/We request the grant of a patent on the basis of this application.

Signature(s)

*Murgitroyd & Co.* Date 23/12/04

12. Name, daytime telephone number and e-mail address, if any, of person to contact in the United Kingdom

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1     **METHOD AND APPARATUS FOR MOVING A FLUID**

2  
3     This invention relates to a method and apparatus for  
4     moving a fluid.

5  
6     The present invention has reference to improvements  
7     to a fluid mover having a number of practical  
8     applications of diverse nature ranging from marine  
9     propulsion systems to pumping applications for  
10    moving and/or mixing fluids and/or solids of the  
11    same or different characteristics. The present  
12    invention also has relevance in the fields inter  
13    alia of heating, cooking, cleaning, aeration, gas  
14    fluidisation, and agitation of fluids and  
15    fluids/solids mixtures, particle separation,  
16    classification, disintegration, mixing,  
17    emulsification, homogenisation, dispersion,  
18    maceration, hydration, atomisation, droplet  
19    production, viscosity reduction, dilution, shear  
20    thinning, transport of thixotropic fluids and  
21    pasteurisation.

1

2 More particularly the invention is concerned with  
3 the provision of an improved fluid mover having  
4 essentially no moving parts.

5

6 Ejectors are well known in the art for moving  
7 working or process fluids by the use of either a  
8 central or an annular jet which emits steam into a  
9 duct in order to move the fluids through or out of  
10 appropriate ducting or into or through another body  
11 of fluid. The ejector principally operates on the  
12 basis of inducing flow by creating negative  
13 pressure, generally by the use of the venturi  
14 principle. The majority of these systems utilise a  
15 central steam nozzle where the induced fluid  
16 generally enters the duct orthogonally to the axis  
17 of the jet, although there are exceptions where the  
18 reverse arrangement is provided. The steam jet is  
19 accelerated through an expansion nozzle into a  
20 mixing chamber where it impinges on and is mixed  
21 with working fluid. The mixture of working fluid  
22 and steam is accelerated to higher velocities within  
23 a downstream convergent section prior to a divergent  
24 section, e.g. a venturi. The pressure gradient  
25 generated in the venturi induces new working fluid  
26 to enter the mixing chamber. The energy transfer  
27 mechanism in most steam ejector systems is a  
28 combination of momentum, heat and mass transfer but  
29 by varying proportions. Many of these systems  
30 employ the momentum transfer associated with a  
31 converging flow, while others involve the generation  
32 of a shock wave in the divergent section. One of

1 working fluid passing through the centre of the  
2 hollow body.

3  
4 PCT/GB2003/004400 describes that the transport fluid  
5 is preferably a condensable fluid and may be a gas  
6 or vapour, for example steam, which may be  
7 introduced in either a continuous or discontinuous  
8 manner. At or near the point of introduction of the  
9 transport fluid, for example immediately downstream  
10 thereof, a pseudo-vena contracta or pseudo  
11 convergent/divergent section is generated, akin to  
12 the convergent/divergent section of conventional  
13 steam ejectors but without the physical constraints  
14 associated therewith since the relevant section is  
15 formed by the effect of the steam impacting upon the  
16 working or process fluid. Accordingly the fluid  
17 mover is more versatile than conventional ejectors  
18 by virtue of a flexible fluidic internal boundary  
19 described by the pseudo-vena contracta. The  
20 flexible boundary lies between the working fluid at  
21 the centre and the solid wall of the unit, and  
22 allows disturbances or pressure fluctuations in the  
23 multi phase flow to be accommodated better than for  
24 a solid wall. This advantageously reduces the  
25 supersonic velocity within the multi phase flow,  
26 resulting in better droplet dispersion, increasing  
27 the momentum transfer zone length, thus producing a  
28 more intense condensation shock wave.

29

30 PCT/GB2003/004400 further discloses that the  
31 positioning and intensity of the shock wave is  
32 variable and controllable depending upon the



1 specific requirements of the system in which the  
2 fluid mover is disposed. The mechanism relies on a  
3 combination of effects in order to achieve its high  
4 versatility and performance, notably heat, momentum  
5 and mass transfer which gives rise to the generation  
6 of the shock wave and also provides for shearing of  
7 the working fluid flow on a continuous basis by  
8 shear dispersion and/or dissociation. Preferably  
9 the nozzle is located as close as possible to the  
10 projected surface of the working fluid in practice  
11 and in this respect a knife edge separation between  
12 the transport fluid or steam and the working fluid  
13 stream is of advantage in order to achieve the  
14 requisite degree of interaction. The angular  
15 orientation of the nozzle with respect to the  
16 working fluid stream is of importance and may be  
17 shallow.

18

19 Further, PCT/GB2003/004400 discloses that the or  
20 each transport fluid nozzle may be of a convergent-  
21 divergent geometry internally thereof, and in  
22 practice the nozzle is configured to give the  
23 supersonic flow of transport fluid within the  
24 passage. For a given steam condition, i.e. dryness,  
25 pressure and temperature, the nozzle is preferably  
26 configured to provide the highest velocity steam  
27 jet, the lowest total pressure drop and the highest  
28 static enthalpy between the steam chamber and the  
29 nozzle exit. The nozzle is preferably configured to  
30 avoid any shock in the nozzle itself. For example  
31 only, and not by way of limitation, an optimum area  
32 ratio for the nozzle, namely exit area: throat area,

1 lies in the range 1.75 and 7.5, with an included  
2 angle of less than  $9^\circ$ .

3  
4 The or each nozzle is conveniently angled towards  
5 the working fluid flow since this helps penetration  
6 of the working fluid by the transport fluid, which  
7 may help shear or thermal dispersion of the working  
8 fluid. This may also prevent both kinetic energy  
9 dissipation on the wall of the passage and premature  
10 condensation of the steam at the wall of the  
11 passage, where an adverse temperature differential  
12 prevails. The angular orientation of the nozzles is  
13 selected for optimum performance which is dependent  
14 inter alia on the nozzle orientation and the  
15 internal geometry of the mixing chamber. Further  
16 the angular orientation of the or each nozzle is  
17 selected to control the pseudo-convergent/divergent  
18 profile, the pressure profile within the mixing  
19 chamber, the enthalpy addition and the condensation  
20 shock wave intensity or position in accordance with  
21 the pressure and flow rates required from the fluid  
22 mover. Moreover, the creation of turbulence,  
23 governed inter alia by the angular orientation of  
24 the nozzle, is important to achieve optimum  
25 performance by dispersal of the working fluid to a  
26 vapour-droplet phase in order to increase  
27 acceleration by momentum transfer. This aspect is of  
28 particular importance when the fluid mover is  
29 employed as a pump. For example, and not by way of  
30 limitation, in the present invention it has been  
31 found that an angular orientation for the or each

1 nozzle may lie in the range 0 to 30° with respect to  
2 the flow direction of the working fluid.

3  
4 A series of nozzles with respective mixing chamber  
5 sections associated therewith may be provided  
6 longitudinally of the passage and in this instance  
7 the nozzles may have different angular orientations,  
8 for example decreasing from the first nozzle in a  
9 downstream direction. Each nozzle may have a  
10 different function from the other or others, for  
11 example pumping, mixing, disintegrating, and may be  
12 selectively brought into operation in practice.  
13 Each nozzle may be configured to give the desired  
14 effects upon the working fluid. Further, in a  
15 multi-nozzle system by the introduction of the  
16 transport fluid, for example steam, phased heating  
17 may be achieved. This approach may be desirable to  
18 provide a gradual heating of the working fluid.

19  
20 An object of the present invention is to improve the  
21 performance of the fluid mover by enhancing the  
22 energy transfer mechanism between the high velocity  
23 transport fluid and the working fluid. This  
24 improves the performance of the fluid mover having  
25 essentially no moving parts having an improved  
26 performance than fluid movers currently available in  
27 the absence of any constriction such as is  
28 exemplified in the prior art recited in the  
29 aforementioned patent.

30  
31 According to a first aspect of the present invention  
32 a fluid mover includes a hollow body provided with a

1 straight-through passage of substantially constant  
2 cross section with an inlet at one end of the  
3 passage and an outlet at the other end of the  
4 passage for the entry and discharge respectively of  
5 a working fluid, a nozzle substantially  
6 circumscribing and opening into said passage  
7 intermediate the inlet and outlet ends thereof, an  
8 inlet communicating with the nozzle for the  
9 introduction of a transport fluid, a mixing chamber  
10 being formed within the passage downstream of the  
11 nozzle, the nozzle internal geometry and the bore  
12 profile immediately upstream of the nozzle exit  
13 being so disposed and configured to optimise the  
14 energy transfer between the transport fluid and  
15 working fluid that in use through the introduction  
16 of transport fluid the working fluid or fluids are  
17 atomised to form a dispersed vapour/droplet flow  
18 regime with locally supersonic flow conditions  
19 within a pseudo-vena contracta, resulting in the  
20 creation of a supersonic condensation shock wave  
21 within the downstream mixing chamber by the  
22 condensation of the transport fluid.

23  
24 The transport fluid is preferably a condensable  
25 fluid and may be a gas or vapour, for example steam,  
26 which may be introduced in either a continuous or  
27 discontinuous manner.

28  
29 According to a second aspect of the present  
30 invention a fluid mover of the kind described in our  
31 aforementioned patent application, includes a hollow  
32 body provided with a straight-through passage of

1 substantially constant cross section with an inlet  
2 at one end of the passage and an outlet at the other  
3 end of the passage for the entry and discharge  
4 respectively of a working fluid, a nozzle  
5 substantially circumscribing and opening into said  
6 passage intermediate the inlet and outlet ends  
7 thereof, an inlet communicating with the nozzle for  
8 the introduction of steam, a mixing chamber being  
9 formed within the passage downstream of the nozzle,  
10 the nozzle internal geometry and the bore profile  
11 immediately upstream of the nozzle exit being so  
12 disposed and configured to optimise the energy  
13 transfer between the steam and working fluid that in  
14 use through the introduction of steam the working  
15 fluid or fluids are atomised to form a dispersed  
16 vapour/droplet flow regime with locally supersonic  
17 flow conditions within a pseudo-vena contracta,  
18 resulting in the creation of a supersonic  
19 condensation shock wave within the downstream mixing  
20 chamber by the condensation of the steam.

21  
22 The nozzle may be of a form to correspond with the  
23 shape of the passage and thus for example a circular  
24 passage would advantageously be provided with an  
25 annular nozzle circumscribing it. The term  
26 'annular' as used herein is deemed to embrace any  
27 configuration of nozzle or nozzles that  
28 circumscribes the passage of the fluid mover, and  
29 encompasses circular, irregular, polygonal and  
30 rectilinear shapes of nozzle.  
31

1 The or each nozzle may be of a convergent-divergent  
2 geometry internally thereof, and in practice the  
3 nozzle is configured to give the supersonic flow of  
4 transport fluid within the passage. For a given  
5 steam condition, i.e. dryness, pressure and  
6 temperature, the nozzle is preferably configured to  
7 provide the highest velocity steam jet, the lowest  
8 total pressure drop and the highest enthalpy between  
9 the steam chamber and nozzle exit.  
10  
11 The condensation profile in the mixing chamber  
12 determines the expansion ratio profile across the  
13 nozzle. With relatively low working fluid  
14 temperatures condensation is dominant, and the exit  
15 pressure of the transport fluid nozzle is low. The  
16 exit pressure of the transport fluid nozzle is  
17 higher when the bulk temperature of the working  
18 fluid is higher.  
19  
20 According to a third aspect of the present invention  
21 a method of moving a working fluid includes  
22 presenting a fluid mover to the working fluid,  
23 the mover having a straight-through passage of  
24 substantially constant cross section,  
25 applying a substantially circumscribing stream  
26 of a transport fluid to the passage through an  
27 annular nozzle,  
28 atomising the working fluid to form a dispersed  
29 vapour and droplet flow regime with locally  
30 supersonic flow conditions,

1           generating a supersonic condensation shock wave  
2   within the passage downstream of the nozzle by  
3   condensation of the transport fluid,

4           inducing flow of the working fluid through the  
5   passage from an inlet to an outlet thereof, and

6           modulating the condensation shock wave to vary  
7   the working fluid discharge from the outlet.

8  
9   Preferably the modulating step includes modulating  
10   the intensity of the condensation shock wave.  
11   Alternatively or additionally the modulating step  
12   includes modulating the position of the condensation  
13   shock wave.

14  
15   The bore profile immediately upstream of the nozzle  
16   is preferably configured to encourage working fluid  
17   atomisation. Preferably an instability in working  
18   fluid flow is introduced immediately upstream of the  
19   nozzle.

20  
21   The or each nozzle is preferably optimally  
22   configured to operate with a particular working  
23   fluid, upstream wall contour profile and mixing  
24   chamber geometry. The nozzles, upstream wall  
25   contour profile and mixing chamber combination are  
26   configured to encourage working fluid atomisation  
27   creating a vapour/droplet mixed flow with local  
28   supersonic flow conditions. This encourages the  
29   formation of the downstream condensation shock wave,  
30   by enhancing local turbulence, pressure gradient and  
31   the momentum and heat transfer rate between the

1 transport and working fluids by maximising surface  
2 contact between the fluids.

3  
4 The or each nozzle is preferably configured to  
5 operate with a particular working fluid, upstream  
6 wall contour profile and mixing chamber to provide  
7 an optimum nozzle exit pressure. Initial pressure  
8 recovery due to transport fluid deceleration,  
9 coupled with the downstream pressure drop due to  
10 condensation, is used to ensure the nozzle expansion  
11 ratio is adjusted to enhance atomisation of the  
12 working fluid and momentum transfer.

13  
14 The exit velocity from the or each nozzle may be  
15 controlled by varying the transport fluid supply  
16 pressure, the expansion ratio of the nozzle and the  
17 condensation profile in the immediate region of the  
18 mixing chamber. The nozzle exit velocities may be  
19 controlled to enhance Momentum Flux Ratios  $M$  in the  
20 immediate region of the mixing chamber, where  $M$  is  
21 defined by the equation

$$22 \quad M = \frac{(\rho_s \times U_s^2)}{(\rho_f \times U_f^2)}$$

23  
24 where  $\rho$  = Fluid density

25  $U$  = Fluid velocity

26 Subscript s represents transport fluid

27 Subscript f represents working fluid

28  
29 In the present invention it has been found that an  
30 optimum Momentum Flux Ratio  $M$  for the or each nozzle  
31 lies in the range  $2 \leq M \leq 70$ . For example, when using



1 steam as the transport fluid, with a working fluid  
2 with a high water content,  $M$  for the or each nozzle  
3 lies in the range  $5 \leq M \leq 40$ .

4  
5 The or each nozzle is configured to provide the  
6 desired combination of axial, radial and tangential  
7 velocity components. It is a combination of axial,  
8 radial and tangential components which influence the  
9 primary turbulent break-up (atomisation) of the  
10 working fluid flow and the pressure gradient.

11  
12 The interaction between the transport fluid and the  
13 working fluid, leading to the atomisation of the  
14 working fluid, is enhanced by flow instability.  
15 Instability enhances the droplet stripping from the  
16 contact surface of the core flow of the working  
17 fluid. A turbulent dissipation layer between the  
18 transport and working fluids is both fluidically and  
19 mechanically (geometry) encouraged ensuring rapid  
20 fluid core dissipation. The pseudo-vena contracta  
21 is a resultant aspect of this droplet atomisation  
22 region.

23  
24 The internal walls of the flow passage upstream of  
25 the or each nozzle may be contoured to provide a  
26 combination of axial, radial and tangential velocity  
27 components of the outer surface of the working fluid  
28 core when it comes into contact with the transport  
29 fluid. It is a combination of these velocity  
30 components which inter alia influence the primary  
31 turbulent break-up (atomisation) of the working

1 fluid and the pressure gradient when it comes into  
2 contact with the transport fluid.

3  
4 Under optimum operating conditions the  
5 disintegration or atomisation of the working fluid  
6 core is extremely rapid. The disintegration across  
7 the whole bore will typically take place in the  
8 mixing chamber within, but not limited to, a  
9 distance approximately equivalent to  $0.66D$   
10 downstream of the nozzle exit. Under different non-  
11 optimised operating conditions disintegration across  
12 the whole bore of the mixing chamber, may still  
13 occur within, but not limited to, a distance  
14 equivalent to  $1.5D$  downstream of the nozzle exit,  
15 where  $D$  is the nominal diameter of the bore through  
16 the centre of the fluid mover.

17  
18 Recirculation occurs in the flow. The  
19 recirculation is particularly dominant where  
20 tangential velocity components of the transport  
21 fluid are present. The radial pressure gradients  
22 created within the mixing chamber are responsible  
23 for this flow phenomenon which encourages complete  
24 and rapid flow dispersion characteristics across the  
25 bore.

26  
27 This effect is also created when the pseudo-vena  
28 contracta is partially established, i.e. vapour-  
29 droplet flow is dominant along the mixing chamber  
30 boundary. The localised pressure gradient draws  
31 flow outwards, causing a region downstream of the  
32 transport fluid nozzle exit, typically between 1

1 diameter and 2 diameters downstream, where the axial  
2 flow component of the working fluid stagnates and  
3 may even reverse briefly on the centre-line, i.e.  
4 the centre of the flow region.

5

6 Recirculation has particular benefits in some  
7 applications such as emulsification.

8

9 A series of nozzles with respective mixing chamber  
10 sections associated therewith may be provided  
11 longitudinally of the passage and in this instance  
12 the nozzles may have different angular orientations,  
13 for example decreasing from the first nozzle in a  
14 downstream direction. Each nozzle may have a  
15 different function from the other or others, for  
16 example pumping, mixing, disintegrating or  
17 emulsifying, and may be selectively brought into  
18 operation in practice. Each nozzle may be  
19 configured to give the desired effects upon the  
20 working fluid. Further, in a multi-nozzle system by  
21 the introduction of the transport fluid, for example  
22 steam, phased heating may be achieved. This  
23 approach may be desirable to provide a gradual  
24 heating of the working fluid, enhanced atomisation,  
25 pressure gradient profiling or a combinatory effect,  
26 such as enhanced emulsification.

27

28 In addition the internal walls of the flow passage  
29 immediately upstream of the or each nozzle exit may  
30 be contoured to provide different degrees of  
31 turbulence to the working fluid prior to its

1 interaction with the transport fluid issuing from  
2 the or each nozzle.

3  
4 The mixing chamber geometry is determined by the  
5 desired and projected output performance and to  
6 match the designed transport fluid conditions and  
7 nozzle geometry. In this respect it will be  
8 appreciated that there is a combinatory effect as  
9 between the various geometric features and their  
10 effect on performance, namely there is interaction  
11 between the various design and performance  
12 parameters having due regard to the defined function  
13 of the fluid mover.

14  
15 According to a fourth aspect of the present  
16 invention a method of processing a working fluid  
17 includes

18 presenting a fluid mover to the working fluid,  
19 the fluid mover having a straight-through passage of  
20 substantially constant cross section,

21 applying a substantially circumscribing stream  
22 of a transport fluid to the passage through an  
23 annular nozzle,

24 atomising the working fluid to form a dispersed  
25 vapour and droplet flow regime with locally  
26 supersonic flow conditions,

27 generating a supersonic condensation shock wave  
28 within the passage downstream of the nozzle by  
29 condensation of the transport fluid, the position of  
30 the condensation shock wave remaining substantially  
31 constant under equilibrium flow,

1 inducing flow of the working fluid through the  
2 passage from an inlet to an outlet thereof, and  
3 varying at least one of a group of parameters  
4 to change the position of the condensation shock  
5 wave, the group of parameters including the inlet  
6 temperature of the working fluid, the flow rate of  
7 the working fluid, the inlet pressure of the working  
8 fluid, the outlet pressure of the working fluid, the  
9 flow rate of a fluid additive added to the working  
10 fluid, the inlet pressure of a fluid additive added  
11 to the working fluid, the outlet pressure of a fluid  
12 additive added to the working fluid, the temperature  
13 of a fluid additive added to the working fluid, the  
14 angle of entry of the transport fluid to the  
15 passage, the inlet temperature of the transport  
16 fluid, the flow rate of the transport fluid, the  
17 inlet pressure of the transport fluid, the internal  
18 dimensions of the passage downstream of the nozzle,  
19 and the internal dimensions of the passage upstream  
20 of the nozzle.

21  
22 The term straight-through when used to describe a  
23 passage encompasses any passage having a clear flow  
24 path therethrough, including curved passages.

25  
26 The fluid additive may be gaseous or liquid. The  
27 fluid additive is not an essential element of the  
28 invention, but in certain circumstances may be  
29 beneficial. The fluid additive may comprise a  
30 powder in dry form or suspended in a fluid.

31

1 The parameter varying step may include switching  
2 between a plurality of transport fluids or between a  
3 plurality of fluid additives.  
4

5 The improvements of the present invention may be  
6 employed to the fluid mover of the aforementioned  
7 patent, and enhance its use in a variety of  
8 applications as disclosed in the aforementioned  
9 patent. These applications range from use as a  
10 fluid processor, including pumping, mixing, heating,  
11 homogenising etc, to marine propulsion, where the  
12 mover is submersed within a body of fluid, namely  
13 the sea or lake or other body of water. In its  
14 application to fluid processing a variety of working  
15 fluids may be processed and may include liquids,  
16 liquids with solids in suspension, slurries, sludges  
17 and the like. It is an advantage of the straight-  
18 through passage of the mover that it can accommodate  
19 material that might find its way into the passage.  
20

21 The fluid mover of the present invention may also be  
22 used for enhanced mixing, dispersion or hydration  
23 and again the combination of the shearing mechanism,  
24 droplet formation and presence of the condensation  
25 shock wave provides the mechanism for achieving the  
26 desired result. In this connection the fluid mover  
27 may be used for mixing one or more fluids, one or  
28 more fluids and solids in particulate form, for  
29 example powders. The fluids may be in liquid or  
30 gaseous form. It has been found that the use of the  
31 present invention when mixing liquid with a powder  
32 of particulate form results in a homogeneous

1 mixture, even when the powder is of material which  
2 is difficult to wet, for example Gum Tragacanth  
3 which is a thickening agent.

4  
5 The treatment of the working fluid, for example  
6 heating, dosing, mixing, dispersing, emulsifying etc  
7 may occur in batch mode using at least one fluid  
8 mover or by way in an in-line or continuous  
9 configuration using one or more fluid movers as  
10 required.

11  
12 A further use to which the present invention may be  
13 put is that of emulsification which is the formation  
14 of a suspension by mixing two or more liquids which  
15 are not soluble in each other, namely small droplets  
16 of one liquid (inner phase) are suspended in the  
17 other liquid(s) (outer phase). Emulsification may  
18 be achieved in the absence of surfactant blends,  
19 although they may be used if so desired. In  
20 addition, due to the straight through nature of the  
21 invention, there is no limitation on the particle  
22 size that can be handled, allowing particle sizes up  
23 to the bore size of the unit to pass through whilst  
24 emulsification is taking place.

25  
26 The fluid mover may also be employed for  
27 disintegration, for example in the paper industry  
28 for disintegration of paper pulp. A typical example  
29 would be in paper recycling, where waste paper or  
30 broken pieces are mixed with water and passed  
31 through the fluid mover. A combination of the heat  
32 addition, the high intensity shearing mechanism, the

1 low pressure region in the vapour-droplet flow and  
2 the condensation shock wave both rapidly hydrates  
3 the paper fibres, and macerates and disintegrates  
4 the paper pieces into smaller sizes. Disintegration  
5 down to individual fibres has been achieved in  
6 tests.

7  
8 The straight through aspect of the invention has the  
9 additional benefit of offering very little flow  
10 restriction and therefore a negligible pressure  
11 drop, when a fluid is moved through it. This is of  
12 particular importance in applications where the  
13 fluid mover is located in a process pipe work and  
14 fluid is pumped through it, such as the case, for  
15 example, when the fluid mover of the present  
16 invention is turned 'off' by the reduction or  
17 stopping of the supply of transport fluid. In  
18 addition, the straight through passage and clear  
19 bore offers no impedance to cleaning 'pigs' or other  
20 similar devices which may be employed to clean the  
21 pipe work.

22  
23 A detailed description of the energy transfer  
24 mechanism, focussing on the momentum transfer  
25 between the transport fluid and working fluid by an  
26 enhanced shearing mechanism is best described with  
27 reference to the accompanying drawings. By way of  
28 example, eight embodiments of geometrical features  
29 that may be employed to enhance this energy transfer  
30 mechanism in accordance with the present invention  
31 are described below with reference to the  
32 accompanying drawings in which:



1

2 Figure 1 is a cross sectional elevation of a fluid  
3 mover according to the present invention;

4 Figure 2 is a magnified view of the shearing  
5 mechanism shown in Figure 1;

6 Figure 3 is a cross sectional elevation of a first  
7 embodiment;

8 Figure 4 is a cross sectional elevation of a second  
9 embodiment;

10 Figure 5 is a cross sectional elevation of a third  
11 embodiment;

12 Figure 6 is a cross sectional elevation of a fourth  
13 embodiment;

14 Figure 7 is a cross sectional elevation of a fifth  
15 embodiment;

16 Figure 8 is a cross sectional elevation of a sixth  
17 embodiment;

18 Figure 9 is a cross sectional elevation of a seventh  
19 embodiment;

20 Figure 10 is a schematic section through the fluid  
21 regime of the fluid mover of the present invention;

22 Figure 11 is a schematic drawing of the fluid mover  
23 of the present invention in use;

24 Figure 12 is a schematic drawing showing pressure in  
25 the fluid mover of the present invention under three  
26 different operating conditions;

27 Figure 13 is a schematic drawing showing a section  
28 through the fluid mover of the present invention and  
29 the pressure distribution in the fluid mover under  
30 two different condensation shock wave positions; and

1 Figures 14a and 14b are partial cross sectional  
2 views through an eighth embodiment of the fluid  
3 mover of the present invention.

4  
5 Like numerals of reference have been used for like  
6 parts throughout the specification.

7  
8 Referring to Figure 1 there is shown a fluid mover  
9 1, comprising a housing 2 defining a passage 3  
10 providing an inlet 4 and an outlet 5, the passage 3  
11 being of substantially constant circular cross  
12 section.

13  
14 The housing 2 contains a plenum 8 for the  
15 introduction of a transport fluid, the plenum 8  
16 being provided with an inlet 10. The distal end of  
17 the plenum is tapered on and defines an annular  
18 nozzle 16. The nozzle 16 being in flow communication  
19 with the plenum 8. The nozzle 16 is so shaped as in  
20 use to give supersonic flow.

21  
22 In operation the inlet 4 is connected to a source of  
23 a process or working fluid. Introduction of the  
24 steam into the fluid mover 1 through the inlet 10  
25 and plenum 8 causes a jet of steam to issue forth  
26 through the nozzle 16. Steam issuing from the  
27 nozzle 16 interacts with the working fluid in a  
28 section of the passage operating as a mixing chamber  
29 (3A). In operation the condensation shock wave 17  
30 is created in the mixing chamber (3A).  
31

1 In operation the steam jet issuing from the nozzle  
2 occasions induction of the working fluid through the  
3 passage 3 which because of its straight through  
4 axial path and lack of any constrictions provides a  
5 substantially constant dimension bore which presents  
6 no obstacle to the flow. At some point determined  
7 by the steam and geometric conditions, and the rate  
8 of heat and mass transfer, the steam condenses  
9 causing a reduction in pressure. The steam  
10 condensation begins shortly before the condensation  
11 shock wave and increases exponentially, ultimately  
12 forming the condensation shock wave 17 itself.

13  
14 The low pressure created shortly before and within  
15 the initial phase of the condensation shock wave  
16 results in a strong fluid induction through the  
17 passage 3. The pressure rises rapidly within and  
18 after the condensation shock wave. The condensation  
19 shock wave therefore represents a distinct pressure  
20 boundary/gradient.

21  
22 The parametric characteristics of the steam coupled  
23 with the geometric features of the nozzle, upstream  
24 wall profile and mixing chamber are selected for  
25 optimum energy transfer from the steam to the  
26 working fluid. The first energy transfer mechanism  
27 is momentum and mass transfer which results in  
28 atomisation of the working fluid. This energy  
29 transfer mechanism is enhanced through turbulence.  
30 Figure 1 shows diagrammatically the break-up, or  
31 atomisation sequence 18 of the working fluid core.

32

1 Figure 2 shows a magnified and exaggerated schematic  
2 of the shearing and atomisation mechanism 18 of the  
3 working fluid by the transport fluid. It is  
4 believed that this mechanism can be broken down into  
5 three distinct regions, each governed by established  
6 turbulence mechanisms. The first region 20  
7 experiences the first interaction between the  
8 transport and working fluid. It is in this region  
9 that Kelvin-Helmholtz instabilities in the surface  
10 contact layer of the working fluid may start to  
11 develop. These instabilities grow due to the shear  
12 conditions, pressure gradients and velocity  
13 fluctuations, leading to Rayleigh-Taylor ligament  
14 break-up 24. Second order eddies within the fluid  
15 surface waves may reduce in size to the scale of  
16 Kolmogorov eddies 22. It is believed that the  
17 formation of these eddies, in association with the  
18 Rayleigh-Taylor ligament break-up, result in the  
19 formation of small droplets 28 of the working fluid.

20  
21 The droplet formation phases may also result in a  
22 localised recirculation zone 26 immediately  
23 following the ligament break-up region. This  
24 recirculation zone may enhance the fluid atomisation  
25 further by re-circulating the larger droplets back  
26 into the high shear region. This recirculation, a  
27 feature of the localised pressure gradient, is  
28 controllable via the transport fluid's axial,  
29 tangential and radial velocity and pressure  
30 components. It is believed that this mechanism  
31 enhances inter alia the mixing, emulsifying and  
32 pumping capabilities of the fluid mover.

1

2 The primary break-up mechanism of the working fluid  
3 core may therefore be enhanced by creating initial  
4 instabilities in the working fluid flow.  
5 Deliberately created instabilities in the transport  
6 fluid/working fluid interaction layer encourage  
7 fluid surface turbulent dissipation resulting in the  
8 working fluid core dispersing into a liquid-ligament  
9 region, followed by a ligament-droplet region where  
10 the ligaments and droplets are still subject to  
11 disintegration due to aerodynamic characteristics.

12

13 Referring now to Figure 3 the fluid mover of Figure  
14 1 and 2 is provided with a contoured internal wall  
15 in the region 19 immediately upstream of the exit of  
16 the steam nozzle 16. The internal wall of the flow  
17 passage 3 immediately upstream of the nozzle 16 is  
18 provided with a tapering wall 30 to provide a  
19 diverging profile leading up to the exit of the  
20 steam nozzle 16. The diverging wall geometry  
21 provides a deceleration of the localised flow,  
22 providing disruption to the boundary layer flow, in  
23 addition to an adverse pressure gradient, which in  
24 turn leads to the generation and propagation of  
25 turbulence in this part of the working fluid flow.  
26 As this turbulence is created immediately prior to  
27 the interaction between the working fluid and the  
28 transport fluid, the instabilities initiated in  
29 these regions enhance the Kelvin-Helmholtz  
30 instabilities and hence ligament and droplet  
31 formation as foreshadowed in the foregoing  
32 description occurs more rapidly.

1  
2 An alternative embodiment is shown in Figure 4.  
3 Again, the fluid mover of Figure 1 and 2 is provided  
4 with a contoured internal wall 19 of the flow  
5 passage 3 immediately upstream of the nozzle 16.  
6 The contoured surface in this embodiment is provided  
7 by a diverging wall 30 on the bore surface leading  
8 up to the exit of the steam nozzle 16, but the taper  
9 is preceded with a step 32. In use, the step  
10 results in a sudden increase in the bore diameter  
11 prior to the tapered section. The step 'trips' the  
12 flow, leading to eddies and turbulent flow in the  
13 working fluid within the diverging section,  
14 immediately prior to its interaction with the steam  
15 issuing from the steam nozzle 16. These eddies  
16 enhance the initial wave instabilities which lead to  
17 ligament formation and rapid fluid cone dispersion.  
18  
19 The tapered diverging section 30 could be tapered  
20 over a range of angles and may be parallel with the  
21 walls of the bore. It is even envisaged that the  
22 tapered section 30 may be tapered to provide a  
23 converging geometry, with the taper reducing to a  
24 diameter at its intersection with the steam nozzle  
25 16 which is preferably not less than the bore  
26 diameter.  
27  
28 The embodiment shown in Figure 4 is illustrated with  
29 the initial step 32 angled at  $90^\circ$  to the axis of the  
30 bore 3. As an alternative to this configuration,  
31 the angle of the step 32 may display a shallower or  
32 greater angle suitable to provide a 'trip' to the

1 flow. Again, the diverging section 30 could be  
2 tapered at different angles and may even be parallel  
3 to the walls of the bore 3. Alternatively, the  
4 tapered section 30 may be tapered to provide a  
5 converging geometry, with the taper reducing to a  
6 diameter at its intersection with the steam nozzle  
7 16 which is preferably not less than the bore  
8 diameter.

9  
10 Figures 5 to 8 illustrate examples of alternative  
11 contoured profiles. All of these are intended to  
12 create turbulence in the working fluid flow  
13 immediately prior to the interaction with the  
14 transport fluid issuing from the nozzle 16.

15  
16 The embodiments illustrated in Figures 5 and 6  
17 incorporate single or multiple triangular cross  
18 section grooves 34, 36 immediately prior to a  
19 tapered or parallel section 30, which is in turn  
20 immediately prior to the exit of the steam nozzle  
21 16.

22  
23 The embodiments illustrated in Figures 7 and 8  
24 incorporate single or multiple triangular 38 and/or  
25 square 40 cross section grooves a short distance  
26 upstream of the exit of the steam nozzle 16. These  
27 embodiments are illustrated without a tapering  
28 diverging section after the grooves.

29  
30 Although Figures 1 to 8 illustrate several  
31 combinations of grooves and tapering sections, it is  
32 envisaged that any combination of these features, or

1 any other groove cross-sectional shape may be  
2 employed.

3

4 The tapered section 30 and/or the step 32 and/or the  
5 grooves 34, 36, 38, 40 may be continuous or  
6 discontinuous in nature around the bore. For  
7 example, a series of tapers and/or grooves and/or  
8 steps may be arranged around the circumference of  
9 the bore in a segmented or 'saw tooth' arrangement.

10

11 The nature of the flow regime in the fluid mover of  
12 the present invention is described in more detail  
13 below, with reference to Figure 10.

14

15 The transport fluid, usually steam 80, enters  
16 through nozzle 16 at supersonic velocity. Wherever  
17 the term stem is used, it is to be understood that  
18 the term can also be applied to other transport  
19 fluids. The working fluid, usually liquid 82, flows  
20 at a subsonic velocity into the inlet 4. At the  
21 nozzle 16 there is a subsonic liquid core 84 which  
22 is bounded by a generally rough or turbulent conical  
23 interface with the steam 80 and the region of  
24 dispersion 88. As the steam 80 exits the nozzle 16  
25 it exhibits local shock and expansion waves 86 and  
26 forms a pseudo vena contracta 90. The accelerated  
27 region of dispersion 88 (or dissociation) of the  
28 liquid core flows at a locally supersonic velocity  
29 into the vapour-droplet region 92, in which the  
30 vapour is steam and the droplets are the working  
31 fluid. Condensation takes place in the supersonic  
32 condensation zone 94 and the subsonic condensation



1 zone 96. The condensation shock wave 17 is produced  
2 when the condensation, which initiates in the  
3 locally supersonic low density region 94, reaches an  
4 exponential rate. The zone 96 immediately after the  
5 condensation shock wave 17 has a considerably higher  
6 density and is hence subsonic. The condensation  
7 shock wave 17 thus defines the interface between  
8 these two densities.

9  
10 In the liquid phase 98 beyond the condensation zone  
11 96 there are small vapour bubbles. The position of  
12 the condensation shock wave is controllable over a  
13 distance L by adjustment of one of the plurality of  
14 parameters described herein.

15  
16 The break-up and dispersion of the primary liquid  
17 core produces a droplet vapour region. Any liquid  
18 instabilities on the primary liquid cone surface 18  
19 are amplified to form 'waves'. These waves are  
20 further elongated to form ligaments that undergo  
21 Rayleigh-Taylor break-up, resulting in the formation  
22 of small droplets 28, separated ligaments 24 and  
23 larger droplets.

24  
25 The secondary region 24 is thus characterised by the  
26 rapid increase in the effective fluid surface area.  
27 These droplets 28, of varying size, are then subject  
28 to several aerodynamic and thermal effects which  
29 ultimately result in their break up to sizes  
30 characteristic with the turbulence levels in this  
31 region. This results in the vapour-droplet region

1     which defines the flow regime within the fluid  
2     mover.

3  
4     The thickness of the viscous sub layer, comprising  
5     the high speed vapour/gas and the locally entrained  
6     liquid in droplet or ligament form, increases  
7     downstream to ultimately extend across the entire  
8     bore. The turbulence within this region arises from  
9     shear (velocity gradient) and eddies (large scale to  
10    Kolmogorov scale), as the flow is essentially of a  
11    vapour-droplet consistency. High levels of shear  
12    exist in the gas/liquid interface.

13  
14    A large amount of energy is transferred in this  
15    secondary region 24 as a result of further particle  
16    break-up. Mass transfer takes place as the shear  
17    forces and thermal discontinuities result in the  
18    droplets becoming ever smaller. The pressure  
19    reduces and droplets are evaporated in order to  
20    maintain equilibrium in the flow. Heat transfer  
21    takes place as equilibrium conditions are reached,  
22    ensuring that liquid vapour phase transitions and  
23    the inverse transitions all occur within the mixing  
24    section of the passage 3. In the secondary region  
25    there is a very rapid increase in the void fraction

$$26 \quad \alpha = \frac{A_g}{A_{Tot}}$$

27  
28    where  $\alpha$  = void fraction

29         $A_g$  = area of gas phase (dispersion cone)

30         $A_{Tot}$  = total area of pump flow

31

1     Thus the rapid increase in specific volume as the  
2     liquid droplets/ligaments are further dispersed,  
3     will obviously result in a larger void fraction.  
4     Subsequently as the flow conditions begin to  
5     approach a state of equilibrium, and due to the  
6     geometry within the mixing chamber, the vapour flow  
7     is encouraged to follow a condensation profile  
8     towards an aerodynamic and condensation shock wave,  
9     which is a region of non-equilibrium and entropy  
10    production.

11  
12    The condensation shock wave arises from the rapid  
13    change from a two-phase fluid mixture to a  
14    substantially single phase fluid with complete  
15    condensation of the vapour phase. Since there is no  
16    unique sonic speed in vapour droplet mixtures, non-  
17    equilibrium and equilibrium exchanges of momentum,  
18    mass and energy can occur. In order to achieve a  
19    normal condensation shock wave, the velocity of the  
20    vapour mixture within the mixing chamber has to be  
21    maintained above a certain value defined as the  
22    equilibrium sonic speed. For conditions where the  
23    vapour velocity is greater than the frozen sonic  
24    speed, or where the velocity of the vapour mixture  
25    is between the equilibrium and frozen sonic speed,  
26    this results in a dispersed or partially dispersed  
27    condensation shock wave. These two asymptotic sonic  
28    speeds are:

29  
30     $a_e$  = equilibrium shock speed. This is the speed at  
31    which every fluid is in its correct equilibrium  
32    condition, i.e. vapour is vapour, liquid is liquid

$a_f$  = frozen shock speed. This occurs primarily due to a 'lag' effect, so that some fluids are not in their correct phase, for example the local temperature and pressure dictate that a vapour should be turning to liquid, but the phase change has not happened.

$a_f$  and  $a_e$  are defined as:

$$a_f = \sqrt{\gamma \cdot R_v \cdot T_s}$$

$$a_e = \sqrt{\frac{\chi \cdot \gamma \cdot R_v \cdot T_s}{\gamma \left[ 1 - \frac{R_v \cdot T_s}{h_{fg}} \left( 2 - \frac{c \cdot T_s}{h_{fg}} \right) \right]}}$$

where

$$c = Cp_v + \frac{\left( \frac{1-\varepsilon}{\varepsilon} \right)}{Cp_f}$$

$\gamma$  = Ratio of specific heats (the vapour and the fluid)

$R_v$  = Gas constant for vapour phase (steam)

$T_s$  = Saturation temperature of mixture (vapour and fluid)

$Cp$  = Specific heat

$H_{fs}$  = Latent heat of vapourisation

$\chi$  = Initial vapour quality

$\varepsilon$  = Vapour fraction (gas/liquid)

Subscript v, represents vapour (steam)

Subscript f, represents fluid (e.g. liquid)

1 Frozen flow arises when the interface transport of  
2 mass, momentum and energy between the vapour phase  
3 and liquid droplets is frozen completely, i.e. the  
4 liquid droplets do not take part in the fluid  
5 mechanical processes.

6  
7 Equilibrium flow arises when the velocity and  
8 temperature of the vapour and liquid are in  
9 equilibrium, and the partial pressure due to the  
10 vapour is equal to the saturation pressure  
11 corresponding to the temperature of the flow.

12  
13 The secondary flow regime can better be understood  
14 by further subdivision into three sub-regions.

15  
16 The first sub-region of the secondary flow regime is  
17 the droplet break-up sub-region. Just as in the  
18 primary zone, where the liquid core is stripped to  
19 form the droplet-vapour zone, with the stripping of  
20 the ligaments and droplets on the surface, so in the  
21 secondary region there is further break-up or  
22 dispersion of these separated ligaments, and also  
23 the break-up of droplets whose characteristics are  
24 unstable in the turbulent flow regime. The dominant  
25 mechanism responsible for the break-up in the  
26 secondary region is the acceleration of droplets or  
27 momentum transfer due to the slip velocity between  
28 vapour and liquid. The injection velocity of the  
29 vapour in the present invention is important to this  
30 functional aspect of the flow regime. If required,  
31 multiple nozzles staggered downstream may be used to  
32 encourage this aspect. Other parameters such as

1 nozzle angle and mixing chamber geometry can be  
2 selected to establish favourable flow conditions.

3  
4 Typical break-up mechanisms in this region are  
5 dependant on the local velocity slip conditions and  
6 the respective working fluid properties. These are  
7 gathered into a dimensionless number referred to as  
8 the aerodynamic Weber number defined as:

$$10 \quad We = \frac{\rho_v \cdot (U_f - U_v)^2 \cdot D_f}{\sigma_f}$$

11

12 where

13  $\rho_v$  = Density of vapour

14  $U$  = Velocity

15  $D_f$  = Hydraulic diameter of fluid

16  $\sigma_f$  = Surface tension of fluid

17

18 Typical break-up mechanisms found in the fluid mover  
19 of the present invention are vibrational break-up,  
20 which can be found with ligaments and droplets whose  
21 characteristic length is greater than the stable  
22 length; catastrophic break-up, which is especially  
23 dominant in the liquid-vapour shear layer where  $We$   
24  $\geq 350$ ; wave crest stripping, which occurs where  
25 droplets, due to their size, experience large  
26 aerodynamic forces causing ellipsoidal shapes,  
27 typically where  $We \geq 300$ ; and short stripping, which  
28 is the dominant break-up mechanism where daughter  
29 and satellite droplets have been formed following  
30 the ligament stripping and dispersion, typically  
31 where  $We \geq 100$ .

1  
2 The turbulent motion of the surrounding gas,  
3 especially where the Reynold numbers are large ( $Re >$   
4  $10^4$ ), as is usually the case in the present  
5 invention, results in large amounts in local energy  
6 dissipation and accompanying droplet break-up. The  
7 fluctuating dynamic pressures resulting from these  
8 turbulent fluctuations are dominant in droplet  
9 break-up but very importantly it is this energy that  
10 ensures extremely effective dispersion and mixing of  
11 the fluids in the flow.

12  
13 Turbulent pressure fluctuations result in shear  
14 forces capable of rupturing fibres or filaments and  
15 dissipating powder lumps or similar solid or semi-  
16 solid matter. In the primary region energy, mass  
17 and momentum transfer takes place through a more  
18 distinct boundary, associated with the liquid cone  
19 dispersion. In the secondary break-up region this  
20 transfer is directly related to the turbulence  
21 intensity, closely associated with the turbulent  
22 dissipation region in the flow.

23  
24 The thermal boundary layer, although similar in  
25 characteristic to the turbulent dissipation  
26 sublayer, represents the effective boundary where  
27 evaporation/condensation and energy transfer occur  
28 in either an equilibrium state or 'frozen' state.

29  
30 Interfacial transport, which begins within the  
31 primary cone dissipation, continues into the  
32 secondary vapour-droplet region and is characterised

1 by distinct mechanisms enhanced within the fluid  
2 mover of the invention through vapour introduction  
3 conditions, dependent on pressure and velocity, the  
4 physical geometry of the steam nozzles and the  
5 mixing chamber geometry. This results in a  
6 continuous surface renewal process, which together  
7 with the turbulence results in a series of renewed  
8 eddies of various scales. These eddies create  
9 bursts arising from the interface of the liquid  
10 vapour and the waves formed on ligaments and  
11 droplets which are undergoing further break-up.  
12 These bursts have a period which is a function of  
13 the interfacial shear velocity. These bursts  
14 greatly encourage mixing, heat transport and  
15 emulsification (droplet size reduction).  
16

17 The second sub-region of the secondary flow regime  
18 is the subcooled vapour-droplet region. As the  
19 vapour mixture flows through the fluid mover of the  
20 invention its velocity profile is adjusted through  
21 fluidic interaction as well as the static pressure  
22 gradient which gradually rises due to general  
23 deceleration of the flow. This controlled diffusion  
24 of the supersonic flow, balance of natural fluidic  
25 and thermodynamic interactions coupled with discrete  
26 geometry results in a vapour-droplet state where  
27 sub-cooled droplets exist within a vapour dominant  
28 phase. The sub-cooled state of this frozen mixture  
29 increases until droplet nucleation, and hence  
30 condensation, begins to occur very rapidly. The  
31 point of maximum sub-cooling (Wilson point)  
32 determines the point at which the nucleation rate,



1 which is closely dependent on sub-cooling because of  
2 the available surface area for condensation, begins  
3 to occur very rapidly, and reaches near exponential  
4 rates. The vapour-droplet region within the fluid  
5 mover of the invention thus is able to attain near  
6 thermodynamic equilibrium within a very short zone.

7  
8 The fluid mover of the invention makes special use  
9 of geometric conditions created through both  
10 geometry and pseudo geometric conditions to ensure  
11 the flow conditions upstream of the critical  
12 subcooled state deviate from the thermodynamic  
13 equilibrium. This ensures maintenance of the  
14 desired vapour-droplet region with its desirable  
15 droplet break-up, particle dispersion and heat  
16 transfer effects.

17  
18 The rapid acceleration of the fluid from the primary  
19 fluid cone into the vapour region results in an  
20 expansion wave, which similarly represents a  
21 thermodynamic discontinuity and allows the vapour  
22 droplet region to deviate markedly from equilibrium  
23 and enter a 'frozen' flow condition.

24  
25 Figure 9 shows an embodiment of the fluid mover of  
26 the invention in which the geometry of the passage 3  
27 has a mixing chamber 3A with a divergent region 50,  
28 a constant diameter region 52 and a re-convergence  
29 profile region 54. The constant through bore is  
30 maintained, but the embodiment of Fig 9 promotes  
31 this expansion and non-equilibrium. This offers

1     excellent particle dispersion, and good flow,  
2     pressure head and suction conditions.

3

4     The third sub-region of the secondary flow regime is  
5     the condensation shock region. As a result of the  
6     sub-cooled vapour-droplet flow regime within the  
7     fluid mover, the point at which exponential  
8     condensation begins to occur defines the  
9     condensation shock wave boundary. The mixture  
10    conditions upstream of the condensation shock wave  
11    determine the nature of the pressure and temperature  
12    recovery experienced within the fluid mover.

13

14    The phase change across the condensation shock wave  
15    obviously results in heat removal from the vapour  
16    phase, although there will be an entropy increase  
17    across the condensation shock wave. The ideal  
18    operating conditions in the fluid mover of the  
19    invention coincide with the formation of a normal  
20    condensation shock wave, referred to as being  
21    discrete, due to its relatively rapid and hence  
22    negligible size measured along the X-axis.

23

24    The nature of the fluid flow in the fluid mover of  
25    the present invention may better be understood by  
26    reference to Figure 12, which shows the distribution  
27    of pressure  $p$  in the fluid mover over length  $x$  along  
28    the axis. Reference is made to the two shock  
29    speeds,  $a_e$  and  $a_f$ , defined earlier.

30

1 Fig. 12a shows condition A and represents the  
2 situation where  $U_{\text{mixture}} > a_e$ , where  $U_{\text{mixture}}$  is the  
3 velocity of the vapour/droplet mixture.

4  
5 This results in a normal condensation shock wave,  
6 with a fairly rapid rise in pressure across the  
7 condensation shock wave. The resulting exit  
8 pressure is higher than the local pressure at the  
9 steam inlet into the bore of the fluid mover.

10  
11 Fig. 12b shows condition B and represents the  
12 situation where  $a_f > U_{\text{mixture}} > a_e$ . In this case the  
13 mixture velocity is higher than the equilibrium  
14 shock speed but less than the frozen shock speed.  
15 In this condition the condensation shock wave is  
16 fully dispersed resulting in a much more gradual  
17 pressure rise across the condensation shock wave.

18  
19 Fig. 12c shows condition C and represents the  
20 situation where  $U_{\text{mixture}} > a_f$ . In this condition an  
21 'unstable' condition arises, with the steam not  
22 fully condensing. This is referred to as a  
23 partially dispersed condensation shock wave. This  
24 results in the start of the formation of a  
25 condensation shock wave (with a reasonably steep  
26 pressure gradient), the condensation shock wave  
27 formation 'stalling', and then restarting again.  
28 However, it has been found that the final resulting  
29 exit pressure is often higher than for either  
30 Condition A or Condition B.

31

1     There are several mechanisms for determining the  
2     state of the flow regime in the fluid mover, and  
3     using this information in a control system to  
4     provide the flow regime that best meets the demands  
5     of the application. For example one can measure the  
6     temperature at a particular point along the length  
7     of the mixing chamber, to determine the existence of  
8     a vapour-droplet region. Such a method is non-  
9     intrusive since the mixer wall can be of thin  
10    section allowing a rapid response to the change in  
11    conditions. Multiple temperature probes spaced  
12    downstream of one another can be used to monitor the  
13    position of the condensation shock wave, as well as  
14    to determine the state of the condensation shock  
15    wave profile.

16  
17    As a further example the use of pressure sensors  
18    allows the condensation shock wave position to be  
19    determined.

20  
21    With reference to Figures 13 and 14 there is shown a  
22    method of using a series of pressure sensors to  
23    detect the position of the condensation shock wave  
24    in the mixing chamber. When the condensation shock  
25    wave 17 is in the position 17A indicated by Case 1,  
26    i.e. in the convergent profile portion 3C of the  
27    passage 3, the pressure profile is shown with the  
28    reference numeral 101. When the condensation shock  
29    wave 17 is in the position 17B indicated by Case 2,  
30    i.e. in the uniform profile portion 3B of the  
31    passage 3, the pressure profile is shown with the  
32    reference numeral 102. Pressure sensors P1, P2 and

1 P3 in the passage 3 can be used to measure the  
2 pressure at three points 103, 104, 105 along the  
3 passage. The pressure measurements at these points  
4 can be used to determine the position of the  
5 condensation shock wave 17. Depending on the flow  
6 profile required, one or more parameters, as  
7 described hereinbefore, can be changed to alter the  
8 flow profile and the position of the condensation  
9 shock wave 17.

10  
11 Figure 14a shows a typical pressure sensor, although  
12 it is to be understood that this is not limiting,  
13 and any suitable pressure sensor or measuring device  
14 may be used. This method of measuring pressures in  
15 the mixing chamber is especially suited for  
16 condensation shock wave detection, since the  
17 measurement technique only needs to measure a change  
18 in pressure rather than being calibrated to measure  
19 accurate values.

20  
21 The mixing chamber 3A is sleeved with a thin walled  
22 inner sleeve 107 of suitable material, such as  
23 stainless steel. A thin layer of oil 108 fills the  
24 gap between the sleeve 107 and the inner wall 106 of  
25 the mixing chamber 3A. The pressure sensor P1 is  
26 located through the wall 106 of the mixing chamber  
27 and is in contact with the oil 108. When the  
28 pressure inside the mixing chamber 3A changes, the  
29 sleeve 107 expands or contracts a small amount,  
30 thereby increasing or decreasing the pressure in the  
31 oil 108, which is then detected by the pressure  
32 sensor P1.

1

2 In the embodiment of Figure 14b the sleeve 107 is  
3 segmented so that the oil is separated by walls 109  
4 fixed to the sleeve. This results in separate  
5 individual chambers of oil 108A, 108B, each with  
6 their own pressure sensor P1, P2. A number of  
7 separate chambers and pressure sensors may be  
8 arranged along the wall 106 of the mixing chamber  
9 3A.

10

11 The advantage of this instrumentation method is that  
12 the sleeve 107 provides a clean inner bore, free of  
13 any crevices or other features in which working  
14 fluid or other transported material can become  
15 trapped. This is of particular relevance for use in  
16 the food industry. In addition, the pressure sensor  
17 P1 is free from contamination, suffers no wear or  
18 abrasion, and does not become blocked.

19

20 A further possible way of monitoring the  
21 condensation shock wave is by the use of acoustic  
22 signatures. Due to the density variation in the  
23 mixer, even during powder addition, it is possible  
24 to determine the 'state' of flow which is an  
25 indication of vapour flow, and hence the condition  
26 of having a condensation shock wave. The mechanisms  
27 for determining the state of the flow regime in the  
28 fluid mover may of course be combined.

29

30 Figure 11 shows an embodiment of the fluid mover 1  
31 with various control means for controlling the  
32 parameters of the flow. The inlet 4 is in fluid

1 communication with a working fluid valve 66 which  
2 can be used to control the flow rate and/or inlet  
3 pressure of the working fluid. A heating means or  
4 cooling means (not shown) may be provided upstream  
5 or downstream of the valve 66 to control the inlet  
6 temperature of the working fluid. The outlet 5 is  
7 in fluid communication with an optional working  
8 fluid outlet valve 68 which can be used to control  
9 the outlet pressure of the working fluid.

10  
11 A transport fluid source 62, such as a steam  
12 generator, is controllable to provide transport  
13 fluid through the transport passage 64 to the plenum  
14 8. The source 62 can be used to control the inlet  
15 temperature and/or the flow rate and/or the inlet  
16 pressure of the transport fluid.

17  
18 The nozzle or nozzles 16 may be mounted for  
19 adjustable movement such that a nozzle angle control  
20 means (not shown) can be used to control the angle  
21 of entry of the transport fluid to the passage.

22  
23 The internal dimensions of the passage downstream of  
24 the nozzle 16 can be adjusted by means of moveable  
25 wall sections 60, which can alter the mixing chamber  
26 wall profile between convergent, parallel and  
27 divergent at a plurality of sections along the  
28 mixing chamber 3A.

29  
30 An additive fluid source 70 may be provided to add  
31 one or more fluids to the working fluid. An  
32 additive fluid valve 72 can be used to control the

1 flow rate of the additive fluid, including to switch  
2 the flow on or off as appropriate. Separate heating  
3 means may be provided for the additive fluid, which  
4 may be a heated liquid, a gas such as steam or a  
5 mixture. The additive may be a powder, and may be  
6 introduced through a valve means from a secondary  
7 hopper.

8  
9 Control means such as a microprocessor may be  
10 provided to control some or all of the parameters  
11 described above as appropriate. The control means  
12 can be linked to the condensation monitoring  
13 devices, such as the pressure sensors P1, P2, P3  
14 which monitor the condensation shock wave, or any  
15 other sensor means eg temperature or acoustic  
16 sensors.

17  
18 The versatility of the fluid mover allows the  
19 present invention to be applied in many different  
20 applications over a wide range of operating  
21 conditions. Furthermore the shape of the fluid  
22 mover of the present invention may be of any  
23 convenient form suitable for the particular  
24 application. Thus the fluid mover of the present  
25 invention may be circular, curvilinear or  
26 rectilinear, to facilitate matching of the fluid  
27 mover to the specific application or size scaling.  
28 The enhancements of the present invention may be  
29 applied to the fluid mover in any of these forms.

30  
31 The fluid mover of the present invention thus has  
32 wide applicability in industries of diverse



1 character ranging from the food industry at one end  
2 of the chain to waste disposal at the other end.

3  
4 The present invention when applied to the fluid  
5 mover of the aforementioned patent affords  
6 particularly enhanced emulsification and  
7 homogenisation capability. Emulsification is also  
8 possible with the deployment of the fluid mover of  
9 the present invention on a once-through basis thus  
10 obviating the need for multi-stage processing. In  
11 this context also the mixing of different liquids  
12 and/or solids is enhanced by virtue of the improved  
13 shearing mechanism which affects the necessary  
14 intimacy between the components being brought  
15 together as exemplified heretofore.

16  
17 The localised turbulence within the working fluid  
18 dispersion region provides rapid mixing, dispersion  
19 and homogenisation of a range of different fluids  
20 and materials, for example powders and oils.

21  
22 The heating of fluids and/or solids can be effected  
23 by the use of the present invention with the fluid  
24 mover by virtue of the use of steam as the transport  
25 fluid and of course in this respect the invention  
26 has multi-capability in terms of being able to pump,  
27 heat, mix and disintegrate etc.

28  
29 The fluid mover of the present invention may be  
30 utilised, for example, in the essence extraction  
31 process such as decaffeination. In this example the  
32 fluid mover may be utilised to pump, heat, entrain,

1 hydrate and intimately mix a wide range of aromatic  
2 materials with a liquid, usually water.

3

4 The vapour-droplet flow region of the present  
5 invention provides a particular advantage for the  
6 hydration of powders. Even extremely hard-to-wet  
7 hydrophilic powders, for example Guar gum, may be  
8 entrained and dispersed into a fluid medium within  
9 this vapour-droplet region.

10

11 As has been disclosed above, the fluid mover of the  
12 present invention possesses a number of advantages  
13 in its operational mode and in the various  
14 applications to which it is relevant. For example  
15 the 'straight-through' nature of the fluid mover  
16 having a substantially constant cross section, with  
17 the bore diameter never reducing to less than the  
18 bore inlet, means that not only will fluids  
19 containing solids be easily handled but also any  
20 rogue material will be swept through the mover  
21 without impedence. The fluid mover of the present  
22 invention is tolerant of a wide range of particulate  
23 sizes and is thus not limited as are conventional  
24 ejectors by the restrictive nature of their physical  
25 convergent sections.



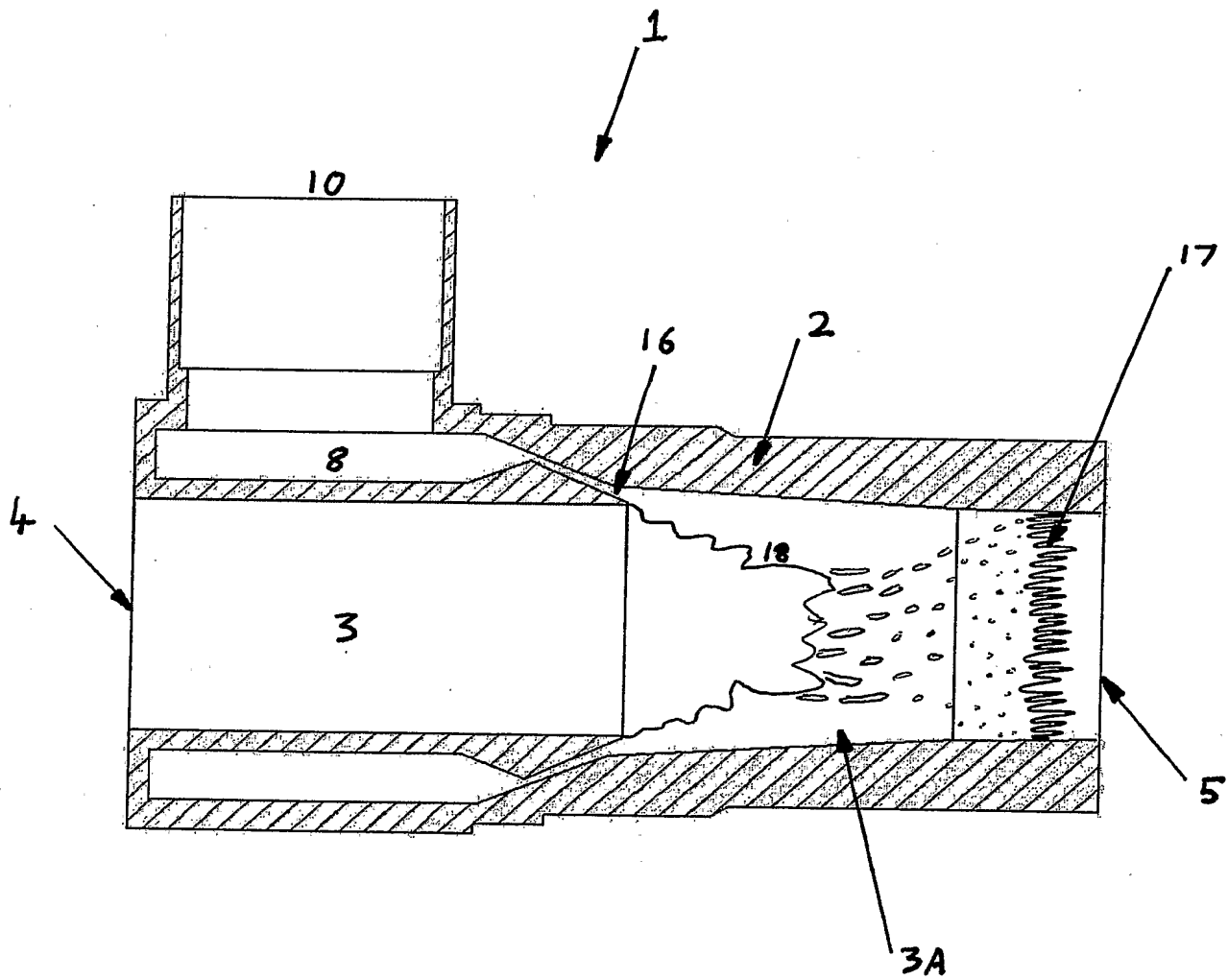


Figure 1



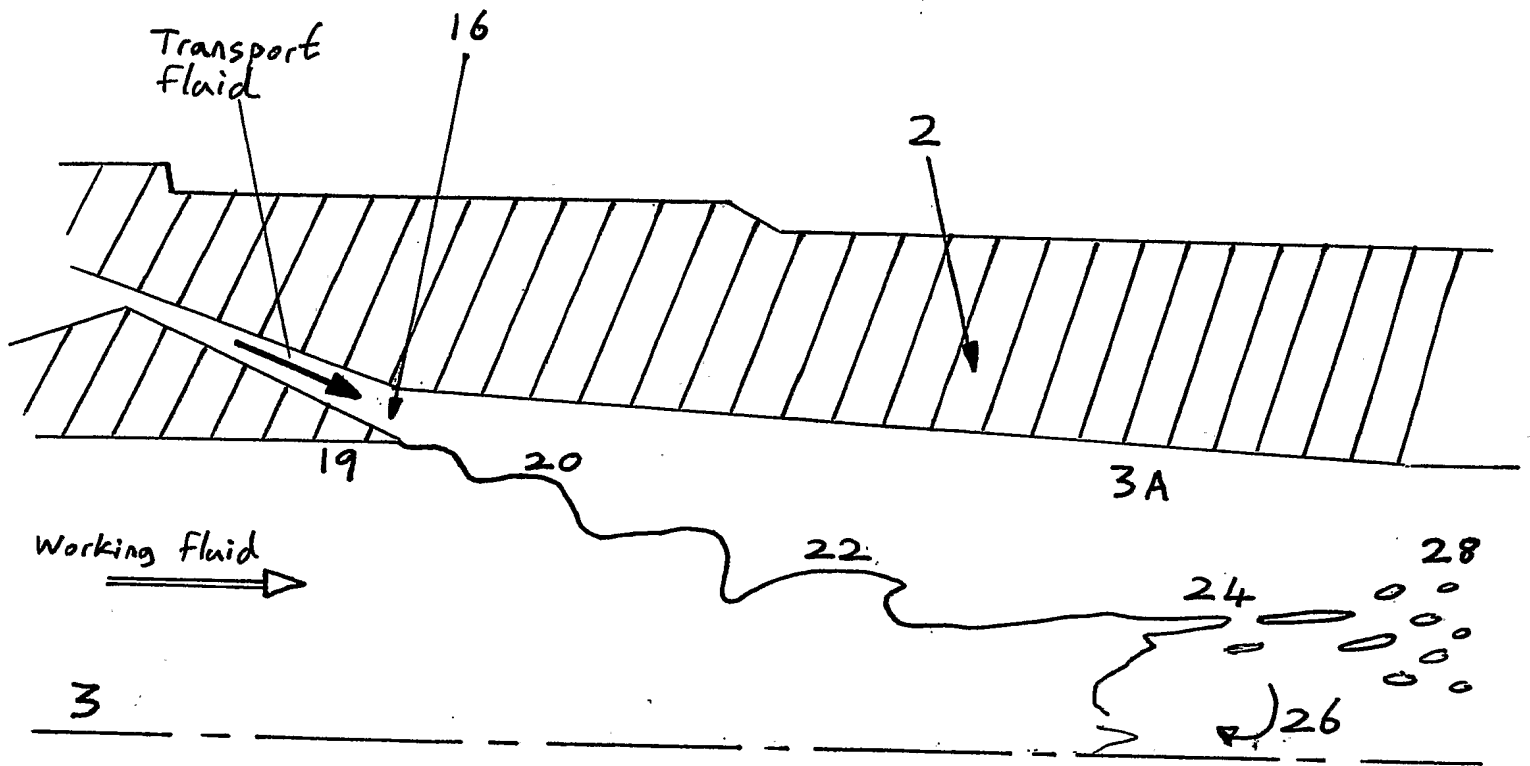


figure 2



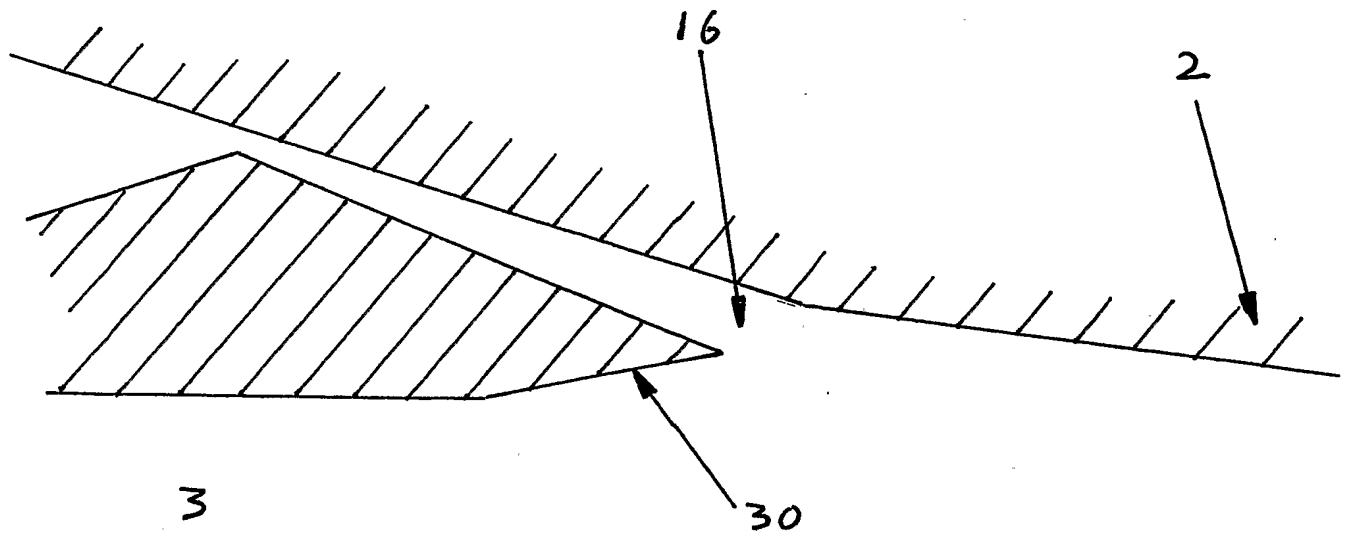


figure 3

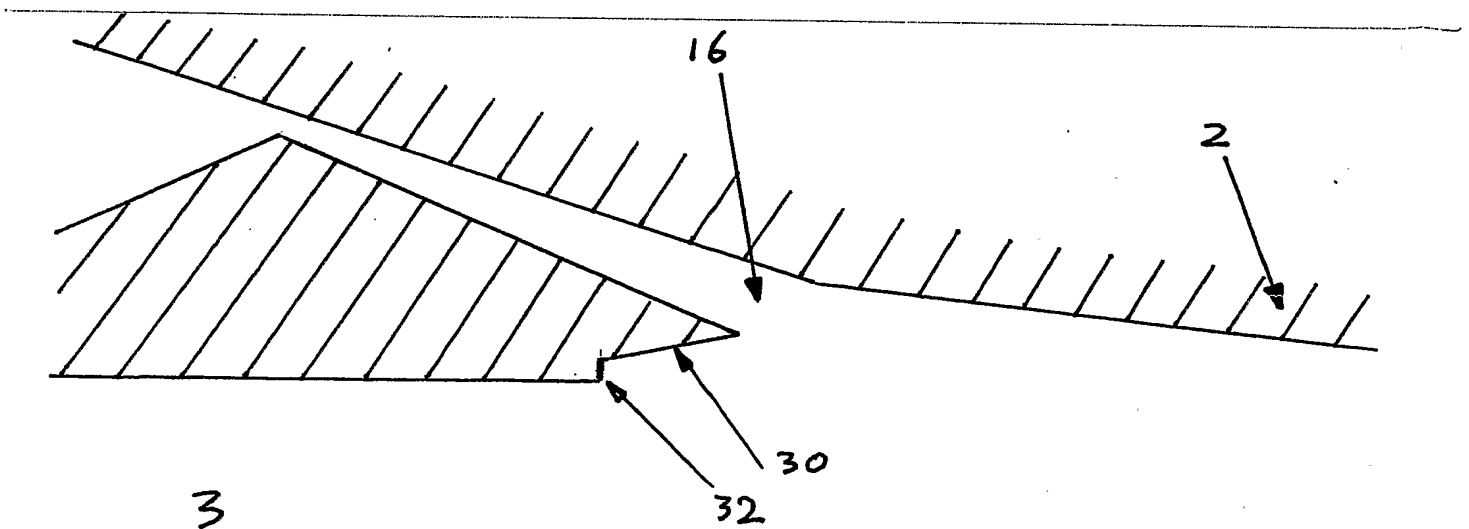
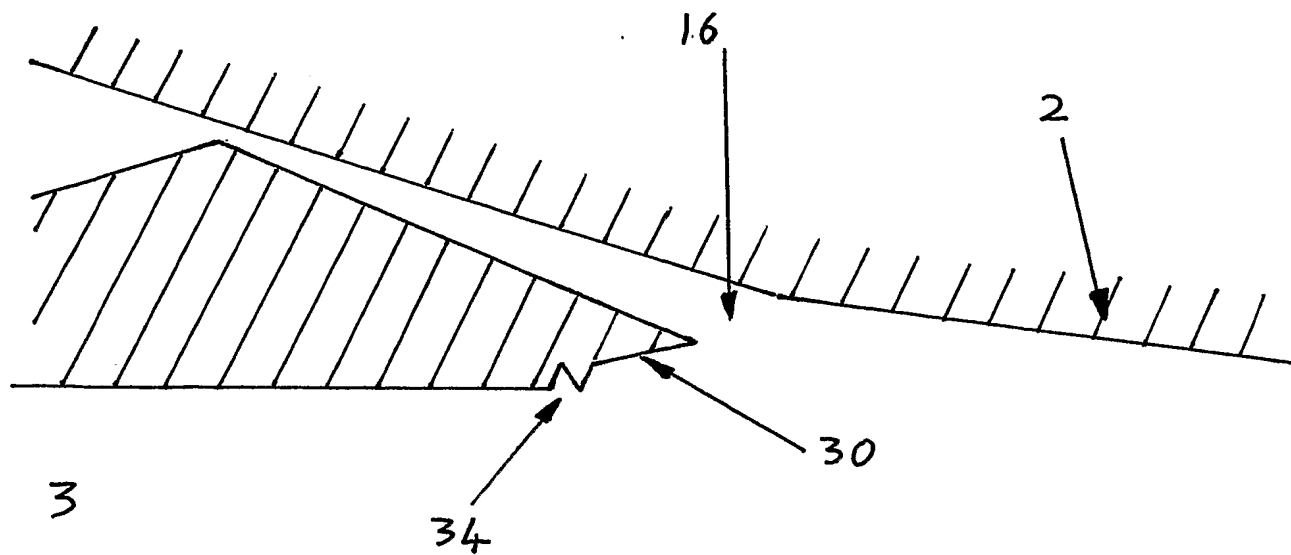
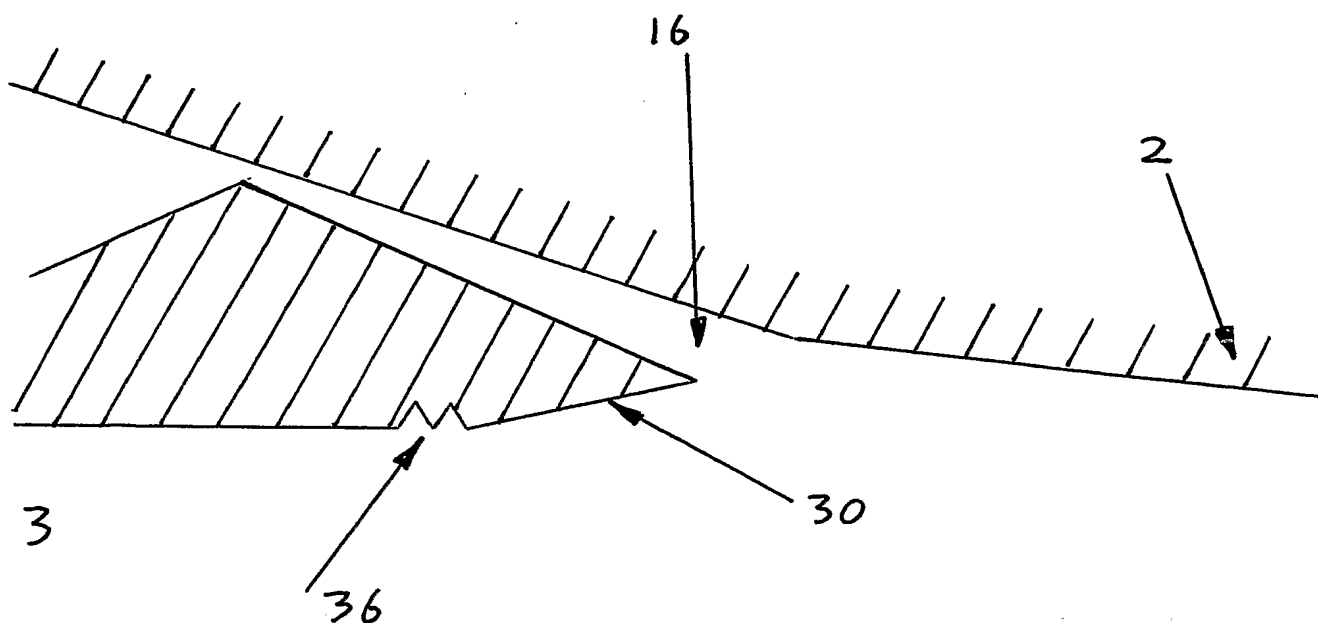


figure 4





figure 5figure 6



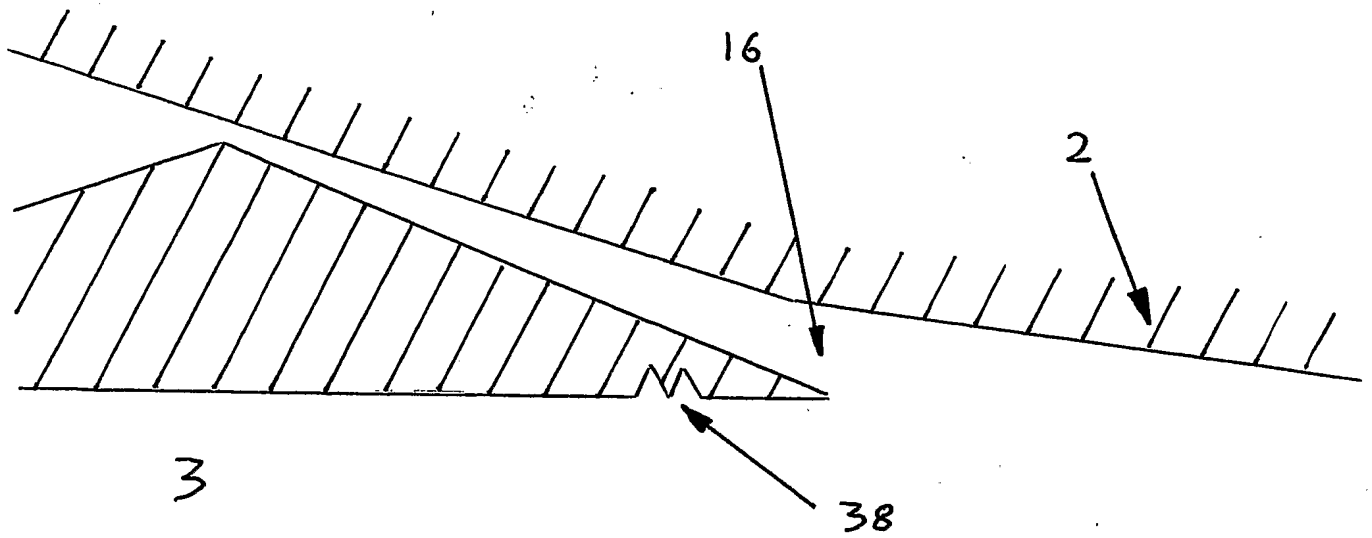


figure 7

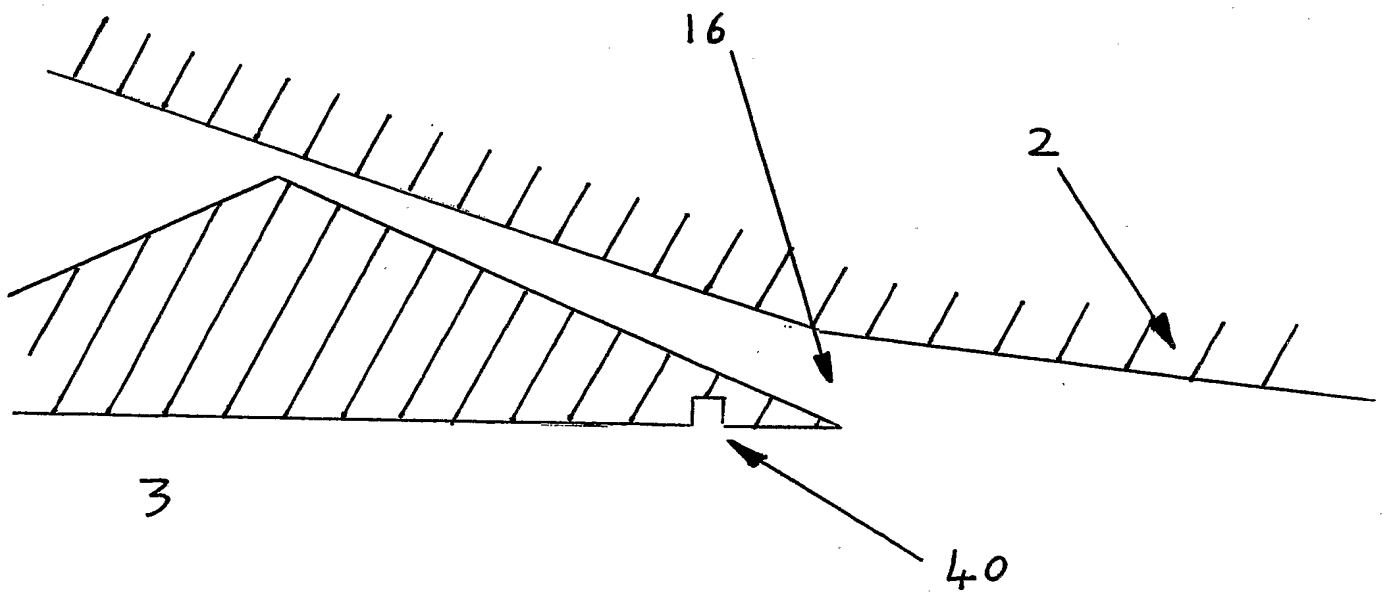


figure 8



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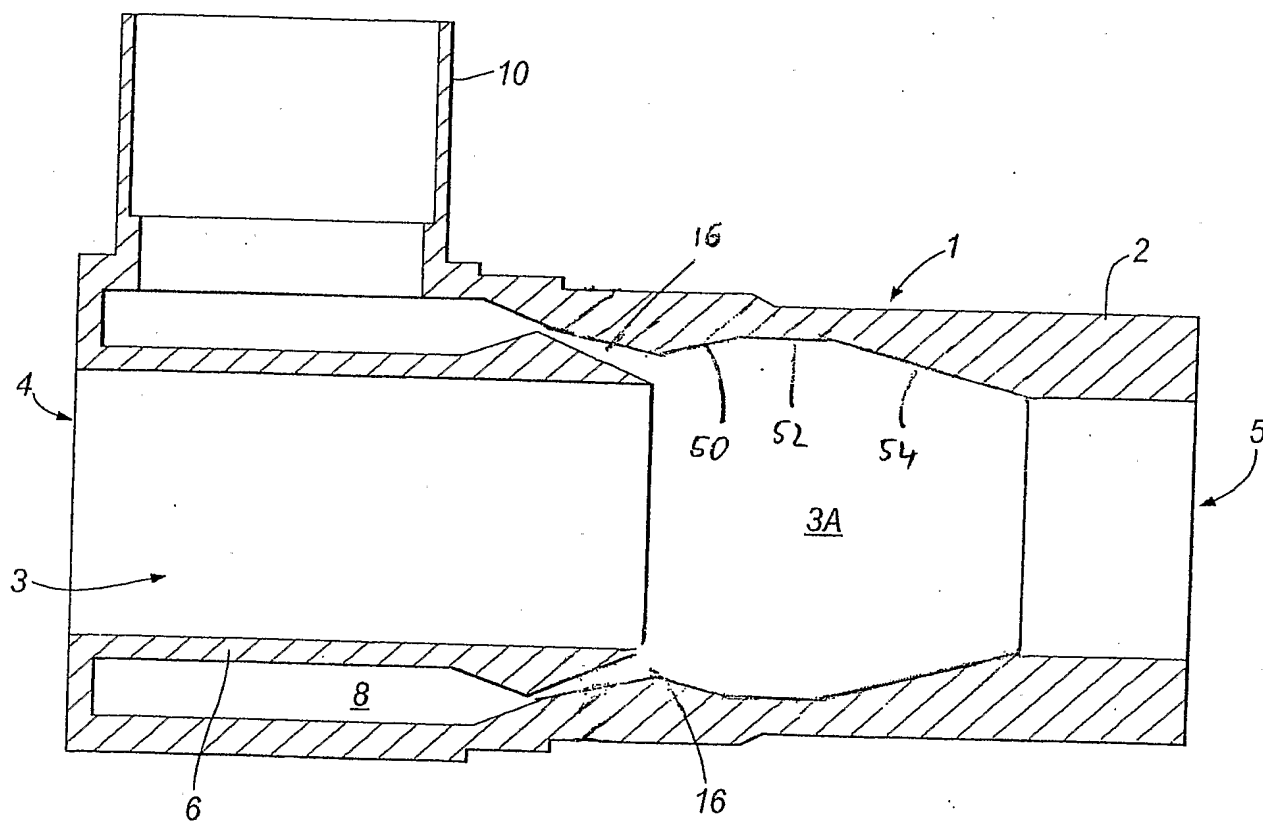


Fig. 9



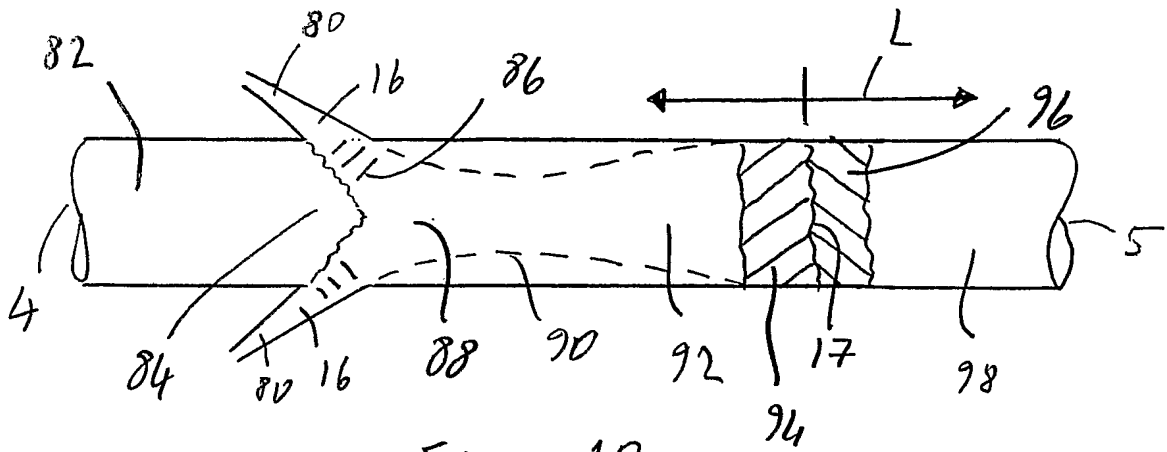


Fig. 10

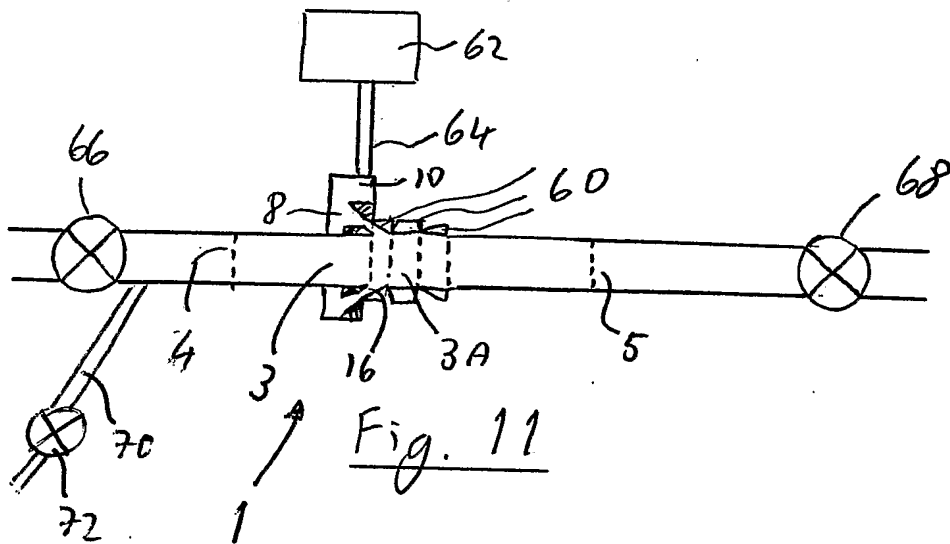


Fig. 11





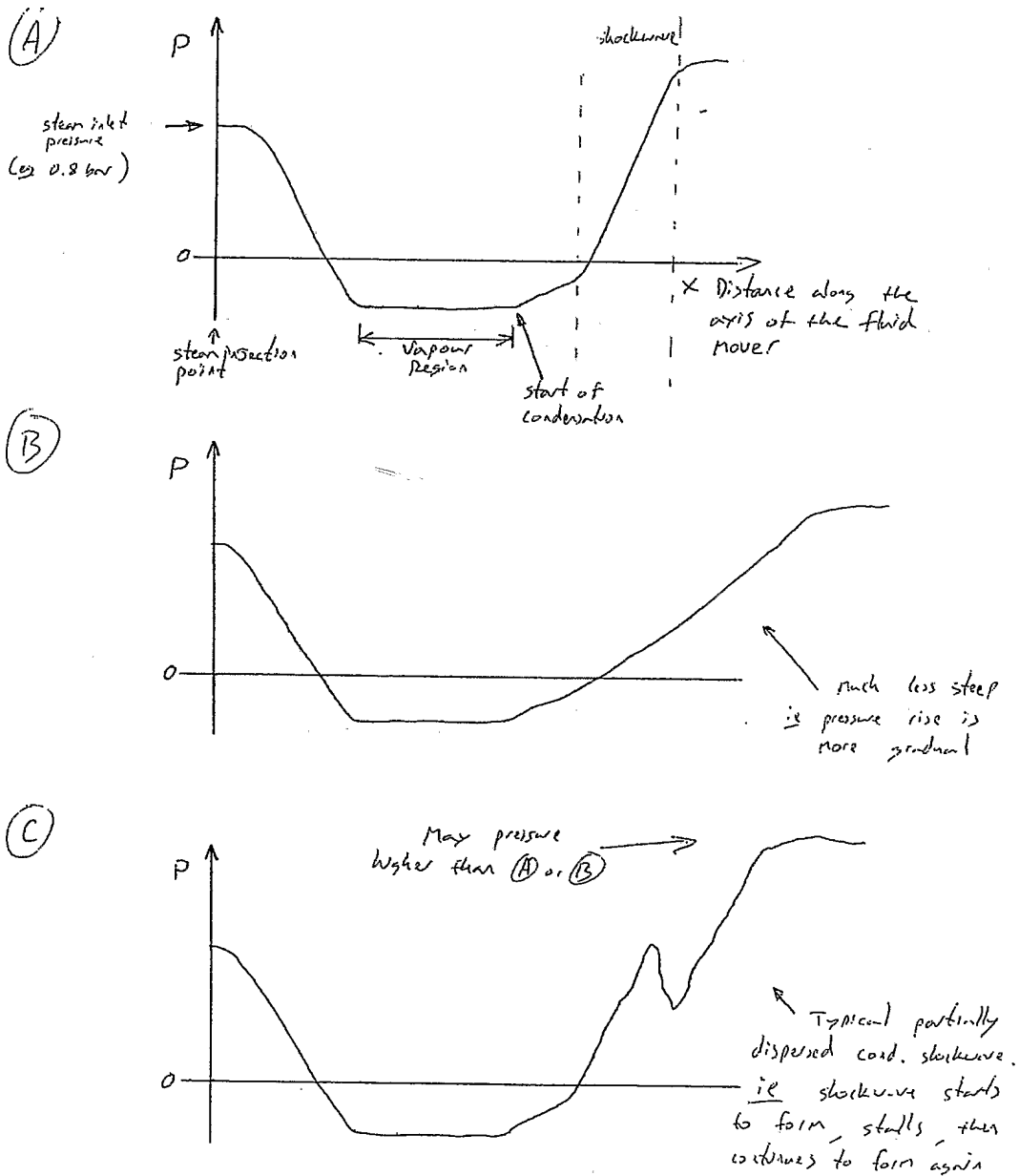
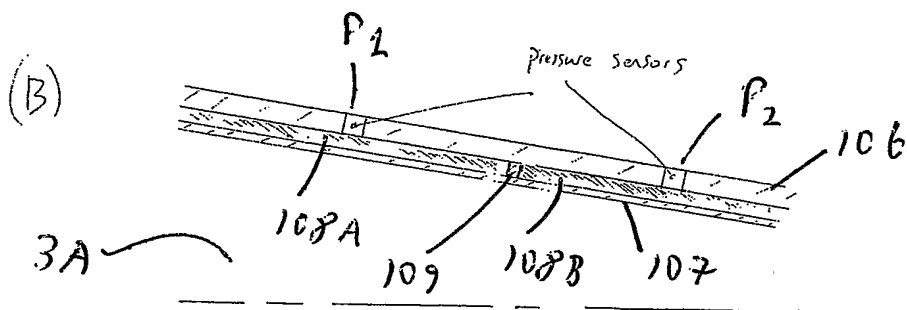
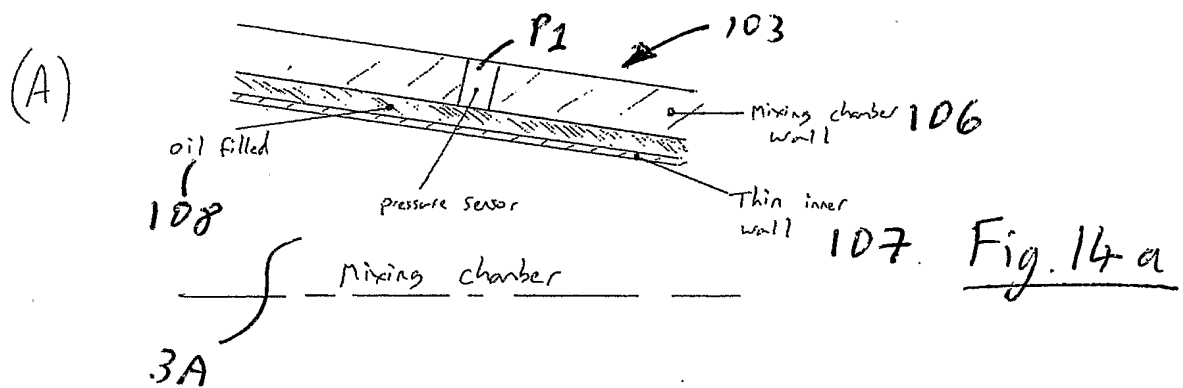
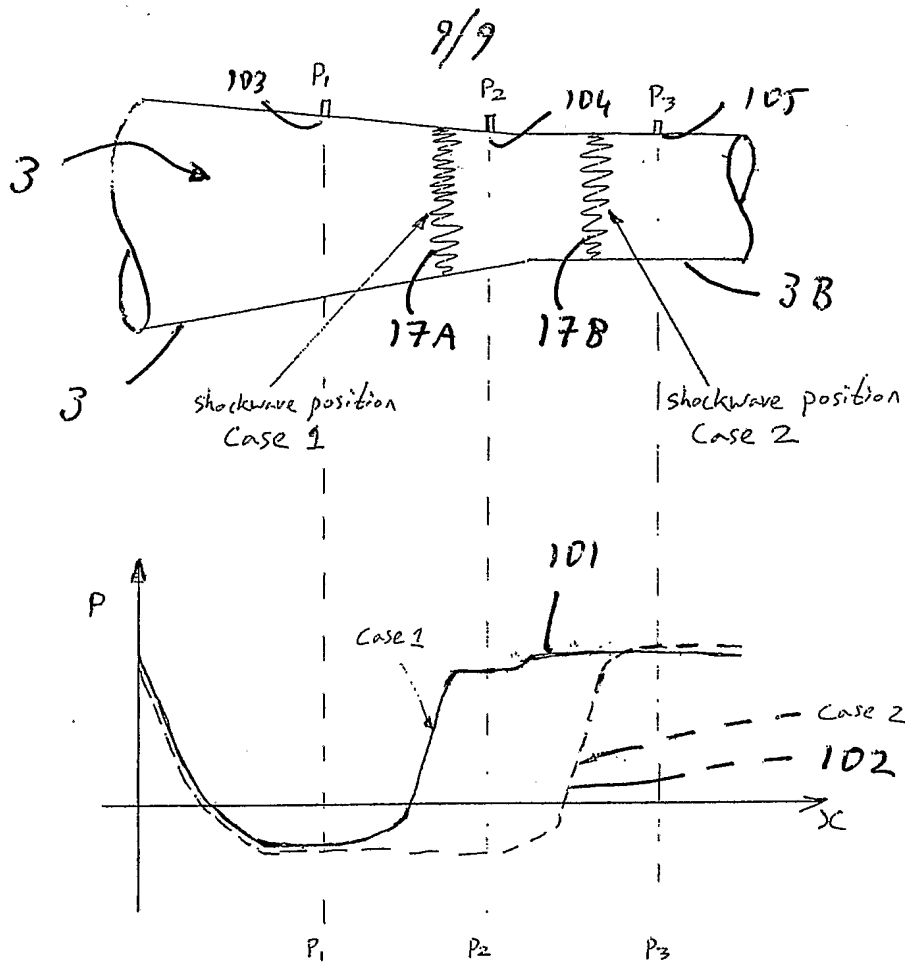


Fig. 12







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